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Krajík, M.; Olesen, Bjarne W.; Petráš, D.

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Sustainable Heating/Cooling for Low Energy Buildings: Experimental Evaluation of Indoor Environment in Residential Rooms with Different Heating/Cooling Concepts

M. Krajčík[1], Bjarne W. Olesen[2] & D. Petráš[1]

[1]Slovak University of Technology, Faculty of Civil Engineering, Radlinského 11, 813 68 Bratislava, Slovakia

[2]ICIEE, Department of Civil Engineering, Technical University of Denmark, Nils Koppels Allé, Building 402 DK-2800 Kgs. Lyngby, Denmark

ABSTRACT

Experimental evaluation is one of the means that allow thorough investigation of the indoor environment in a room. Providing that the measurement procedures are correct and that the investigator has the necessary experimental equipment available, experimental measurements can provide results with high accuracy and under well defined boundary conditions, which can be further verified by field measurements or used for validation of a computer simulation. A set of experimental studies of air distribution, ventilation effectiveness and thermal environment were carried out in a simulated room heated/cooled and ventilated by different concepts, at various boundary conditions, differing in supply air temperature, floor temperature, simulated heat gain/heat loss, nominal air change rate and positions of air terminal devices. The experimental room simulated corresponds to a residential room or a single office room located in a low-energy building. Procedures and indicators that can be successfully used for experimental investigations of indoor environment are described and a sample of measured data is reported.

1. INTRODUCTION

1.1 ENERGY POINT OF VIEW

All-air and radiant systems are two important, but not the only heating/cooling and ventilation solutions for low energy buildings. There are studies indicating that radiant systems outperform all-air systems in terms of energy (Babiak et al., 2007; Fabrizio et al., 2010), whereas other studies suggest that using an all-air system is a viable solution, especially for very low energy houses (Feist et al., 2005). However, in buildings, where the heating demand is very low, the differences in energy consumption between different alternatives of heating/cooling and ventilation systems may become small and other factors than energy consumption may become more important for the choice of suitable HVAC system. One very important factor is the quality of the indoor environment.

1.2 INDOOR ENVIRONMENT POINT OF VIEW

One of the current trends in design and operation of systems for air conditioning of the indoor environment is separation of the functions of creating thermal comfort in winter period (covering heat loss by a heating system), creating thermal comfort in summer period (elimination of heat gains by a cooling system), as well as ensuring the necessary air change by means of natural ventilation, mechanical ventilation or their combination. Air conditioning systems transferring the heat/cold by convection and ventilating the space at the same time are being substituted by systems, where most of the heat is transferred by radiation, coupled by a separate ventilation system supplying the amount of air necessary to fulfil hygiene requirements. Therefore, applications with low temperature radiant heating and high temperature radiant cooling in combination with mixing, displacement or personalized ventilation are considered as progressive systems. For such combined systems the thermal environment and air distribution patterns may be different from when the systems are operated as single. When properly designed, combination of different heating/cooling and ventilation systems can take advantage of the positive characteristics of the respective systems and create a comfortable thermal environment with a very good indoor air quality. However, when not applied properly, using of such systems may lead to thermal discomfort and short-circuit airflow patterns. This trend is opposed by the philosophy of using only one system for both ventilation and heating/cooling at the same time. Such a concept might lead to operation energy savings and lower investment costs, but the potential risk of such systems is represented mainly by the fact that the ventilation effectiveness depends on the position of the supply and extract air terminal devices and on the difference between supply and room air temperature, so when the ventilation air is warmer than the room air, the ventilation effectiveness can be as low as 0.4, where 1 represents complete mixing (CR 1752, 1998). The present paper deals with experimental measurements, which can be employed as a tool to evaluate different ventilation concepts at various boundary conditions and to help obtain information on thermal environment and air distribution patterns created by these systems.

2. INDOOR AIR QUALITY INDICATORS

2.1 THERMAL ENVIRONMENT

2.1.1 Temperature effectiveness

Knowing the air temperature in the occupied zone, in the supply airflow and in the exhaust airflow it is possible to calculate the temperature effectiveness as:

$$\varepsilon_t = (t_e - t_s) / (t_i - t_s) \qquad (1)$$

where t_e is the exhaust air temperature, t_s is the supply air temperature and t_i is the mean air temperature in the occupied zone. With values of ε_t lower than 1 the supply air temperature must be increased when heating in order to obtain the required room temperature. Reported temperature effectiveness refers to 0.6 m above the floor at the occupied zone, representing the human body mass balance of a sitting person. However, the temperature effectiveness is a valid indicator only in case when the room is heated/cooled by the supply air.

2.1.2 Manikin-based equivalent temperature

Manikin-based equivalent temperature (t_{eq}) was used to evaluate possible discomfort due to non-uniform thermal environment and local cooling of certain body parts. It can be interpreted as the temperature that a person would sense on various body parts in the actual environment. A thermal manikin in sitting posture during experimental measurements is shown in Fig.1.

Fig. 1 Thermal manikin simulating internal heat gains and measuring equivalent temperatures.

The manikin can be operated in a comfort equation mode, in which the heat supply to particular body segments is maintained equal to the heat loss from that segment, in order to maintain thermal neutrality. The thermal manikin cannot simulate the latent heat loss, so this value cannot be measured directly. The relation between the total heat loss and sensible heat loss is obtained assuming a vapour pressure of 1.5 kPa, equivalent to typical indoor conditions at 24 ºC and relative humidity of 50 %. The sensible heat loss Q_s can be derived from the power supplied to the body segments. The skin temperatures of the body segments, $t_{sk,i}$, are then calculated using the following equation:

$$t_{sk,i} = 36.4 - 0.054 \cdot Q_{s,i} \quad (2)$$

while t_{eq} is calculated for i body segments as:

$$t_{eq,i} = t_{sk,i} - Q_{s,i} / h_{cal,i} \quad (3)$$

where $h_{cal,i}$ is the heat transfer coefficient for a body segment, determined from calibration in a uniform thermal environment.

2.1.3 Draught rating

Draught rating (DR) can be determined from measured air temperatures, mean air velocities and the standard deviations of air velocities according to EN ISO 7730 (2005):

$$DR = (34 - t_a) \cdot (v - 0.05)^{0.62} \cdot (0.37 \cdot v \cdot Tu + 3.14) \quad (4)$$

where t_a is the local air temperature in °C, v is the local mean air velocity in m/s and Tu is the local turbulence intensity in %.

2.1.4 Contaminant removal effectiveness (CRE)

CRE indicates the ability of the room ventilation system to remove air-borne contaminants when the position of the contaminant source is known. The CRE can be then calculated as:

$$CRE = (c_e - c_s) / (c_i - c_s) \tag{5}$$

where c_e is the contaminant concentration in the exhaust air, c_i is the mean contaminant concentration in the room and c_s is the contaminant concentration in the supply air. At complete mixing the concentration at any point in the room is equal to the concentration in the exhaust airflow and the CRE is equal to 1.

2.1.5 Air change efficiency

ACE is a more general indicator than the CRE and can be used when the position of the contaminant source is not specified. One possible way to express ACE is to compare the local mean age of air at the exhaust (the nominal time constant) with the local mean age of air at a point in the occupied zone. At complete mixing the nominal time constant is the same as the room age of air and the ratio is equal to 1. The is then calculated as:

$$ACE = \tau_n / \tau_p \,.\, 100\ (\%) \tag{6}$$

where τ_n is the local mean age of air at the exhaust, equal to the nominal time constant, and τ_p is the mean age of air at a particular point in the occupied zone. The mean age of air at a certain point can be defined as the mean transit time, i.e. the time it takes for molecules of fresh air to reach the point, plus the mean presence time. There are several methods to determine the age of air, e.g. the tracer step-down (decay) method, pulse method or the tracer step-up method. An example to explain the way the mean transit time and the mean presence time can be determined using the tracer step-up method is shown in Fig.2. In this method the tracer gas is continuously released in the supply air duct at a constant rate. The release of tracer gas starts at time t=0 and the increase in the concentration is continuously recorded at a point in the room and in the exhaust airflow. The mean age of air at a certain point is determined from the tracer gas concentration curves as a function of time, using the equation:

$$\tau_p = \frac{1}{C(t_\infty)} \int_0^{t_\infty} \left[C(t_\infty) - C(t) \right] dt \tag{7}$$

where $C(\infty)$ is the steady state tracer gas concentration and $C(t)$ is the concentration recorded in a step-up experiment.

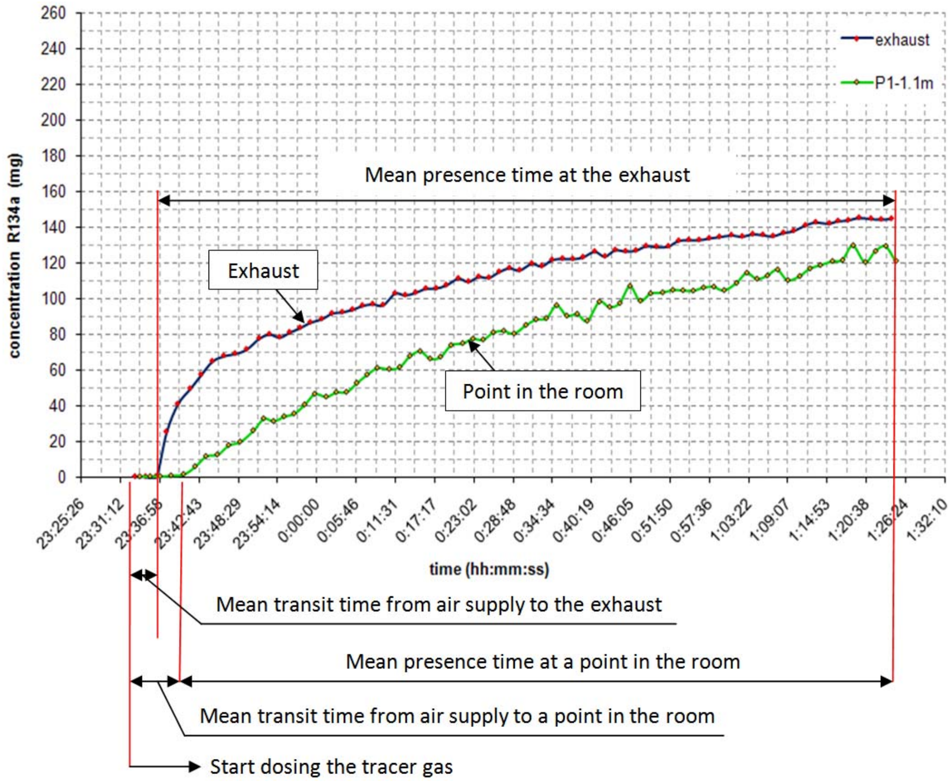

Fig. 2 Determination of the mean transit time and the mean presence time using tracer step-up method.

3. EXPERIMENTAL MEASUREMENTS

3.1.1 Measurement procedure

Measuring positions used in the presented experiment are shown in Fig. 3. The points marked with „S" represent the positions of the air temperature and air velocity sensors, and the positions of the measuring points for ventilation effectiveness are marked with „C". All measurements of air temperature, air velocity and tracer gas concentrations were recorded under steady-state thermal conditions. The layout of the chamber was considered symmetrical and the measuring points were therefore located only in the half of the room with simulated occupancy. The occupied zone was considered to be between the floor and 1.8 m above the floor, about 1 m from the cold window and 0.3 m from the internal walls. During the experiments, the room air temperature was controlled based on air temperature measured at the reference point, located in the middle of the room at 1.1 m above the floor (Fig. 3). The reference air temperature was 22 °C in winter and 26 °C in summer. Heat loss/heat gain was simulated through radiant heating/cooling panels located on one of the walls. In the CRE measurements tracer gas was released at a constant rate on the side of the table opposite the manikin. The simulated contaminant source was located 1.1 m above the floor level. When the tracer gas concentration reached steady state, samples were taken from positions C1, C2 and C3 at 0.6 m, 1.1 m and 1.7 m above the floor, in order to describe the CRE at the occupied zone, and from the exhaust air.

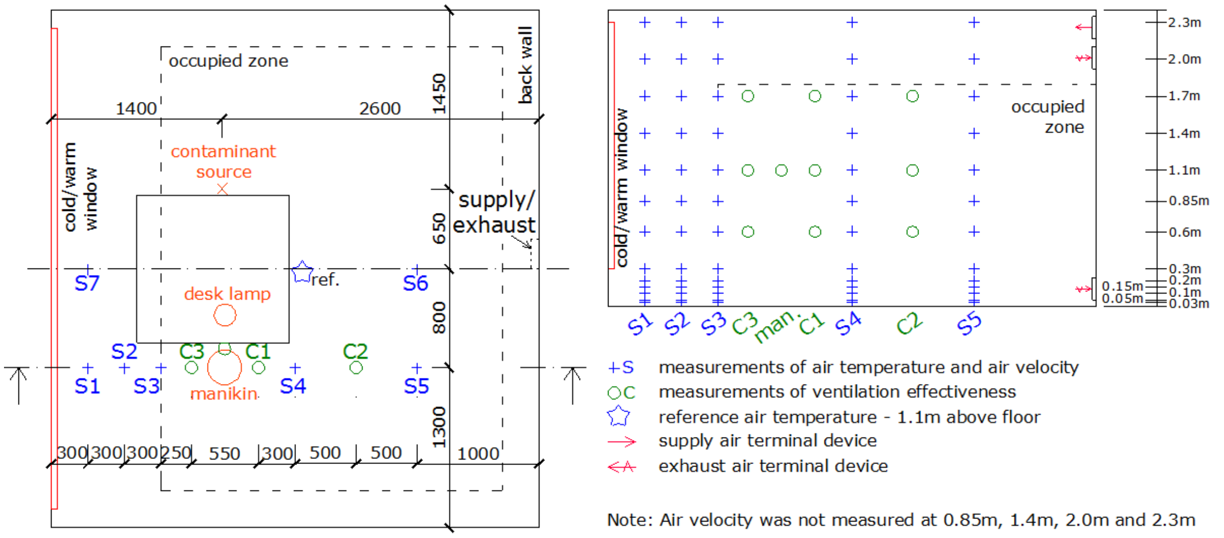

Fig. 3 Plan and cross-section of the experimental room. Measurement positions of air/operative temperature and air velocity are indicated by a cross, measurement positions of ventilation effectiveness are indicated by a circle.

3.1.2 Results for winter conditions

Sample of the results of experimental measurements for winter period, together with simulated conditions, is given in Tab. 1. In System W1 both supply and exhaust terminal were located at the ceiling. In System W2 supply terminal was at the ceiling and exhaust terminal was at the floor level. Cases 1 to 3 represent heating and ventilation by warm air supplied in the room through a ventilation system.

Tab. 1 Experimental conditions and results of the measurements for winter conditions (Krajčík, 2012).

System/ case	Nominal ACH (h^{-1})	Calculated heat loss (W/m^2)	Internal heat gains (W)	Supply air temp. minus room temp. (°C)	Room temp. minus window temp. (°C)	Vertical air temp. difference in occupied zone[b] (°C)	Air temp. effectiveness in occupied zone at 0.6m	CRE Manikin at 1.1m[c]	CRE C1 at 1.1m	CRE Occupied zone[d]	ACE C1 at 1.1m
Syst. W1											
case 1	0.5	13	90	10.3	3.2	0.8	0.8	0.63±0.01	0.70	0.67±0.05	0.42
case 2	1.0	13	90	6.2	3.3	0.8	0.8	0.58±0.01	0.62	0.59±0.03	0.45
case 3	1.5	13	90	6.2	3.2	0.8	0.8	0.62±0	0.77	0.71±0.1	0.60
case 4[a]	0.5	13	90	-6.0	3.2	-0.1	-	0.90±0.01	0.97	0.94±0.03	1.00
Syst. W2											
case 1	0.5	13	88	6.5	3.7	0.7	0.7	0.83±0	0.91	0.86±0.06	1.20
case 2	1.0	13	89	5.5	3.5	0.8	0.8	0.85±0.04	0.87	0.86±0.1	0.97
case 3	1.5	13	89	5.6	3.3	0.8	0.8	0.69±0.01	0.83	0.75±0.1	0.77
case 4[a]	0.5	13	90	-5.7	3.4	0.0	-	0.93±0.01	0.96	0.94±0.03	n/a

a) Combination of mixing ventilation and floor heating.
b) Vertical air temperature difference between 1.7 m and 0.1 m above the floor.
c) Mean of three measurements at one position (manikin´s breathing zone) ± 95 % confidence limit.
d) Mean value from one measurement at nine positions (C1, C2 and C3 at 0.6 m, 1.1m and 1.7m) ± standard deviation.

The three warm air heating cases differ in the nominal air change rate, when the air change of 0.5 h^{-1} represents the minimum air change required to fulfil the hygiene requirements necessary for residential rooms (EN 15251) and the air change of 1 and 1.5 h^{-1} were chosen to investigate the effect of increased air change. Case 4 represents floor heating, assuming a balanced ventilation system with heat recovery supplying ventilation air at the temperature of 16 °C.

At small air change rates, for warm air heating the vertical air temperature difference never exceeded 0.8 °C. Homogenous thermal environment was also confirmed by the equivalent temperatures, when the vertical equivalent temperature difference between head and feet did not exceed 2 °C. The vertical air temperature difference was always about zero when floor heating was in operation. The air movement increased at around 0.1 m above the floor. The air velocity seldom reached 0.1 m/s when the room was heated by warm air and it reached values up to 0.15 m/s for combination of mixing ventilation and floor heating. It follows that the draught rating in the occupied zone was also low and it seldom reached values greater than 10 %.

3.1.3 Results for summer conditions

In Tab. 2 is sample of results of experimental measurements for summer period. In System S1 both supply and exhaust were located on the wall, the supply above the exhaust. In System S2 the supply was located on the wall in the upper part in the room and the exhaust was located on the wall just above the floor level. Cases 1 and 2 represent floor cooling in summer conditions, with supply of fresh warm outside air at 30 °C at 0.5 h^{-1} or 1.0 h^{-1}, respectively. Case 3 represent floor cooling combined with mechanical ventilation supplying cold ventilation air. When floor cooling was combined with warm outdoor air supply, the average vertical air temperature difference in the occupied zone exceeded 4 °C and the air velocity was very low, up to 0.07 m/s. For both floor cooling systems the results are similar regardless the nominal air change rate.

Tab. 2 Experimental conditions and results of the measurements for summer conditions (Krajčík, 2012).

System/ case	Nominal ACH (h^{-1})	Calculated heat loss (W/m^2)	Internal heat gains (W)	Supply air temp. minus room temp. (°C)	Room temp. minus window temp. (°C)	Vertical air temp. difference in occupied zone[a] (°C)	Air temp. effective-ness in occupied zone at 0.6m	CRE Manikin at 1.1m[b]	CRE C1 at 1.1m	CRE Occupied zone[c]	ACE C1 at 1.1m
Syst. S1											
case 1	0.5	25	5.4	4.3	6.6	4.1	0.06	1.15±0.08	0.96	1.10±0.08	0.52
case 2	1.0	24	5.3	4.4	6.2	4.4	0.07	0.92±0.04	1.02	0.94±0.19	0.70
case 3	0.5	32	5.1	-7.1	8.5	2.4	0.06	1.08±0.00	0.99	0.96±0.11	0.95
Syst. S2											
case 1	0.5	23	5.3	3.6	6.1	3.8	0.06	0.78±0.03	0.74	0.74±0.06	1.05
case 2	1.0	21	5.3	3.7	5.4	n/a	0.07	0.97±0.05	0.98	1.00±0.06	1.12
case 3	0.5	25	4.9	-7.0	6.6	1.3	0.06	0.93±0.00	0.85	0.83±0.06	1.01

a) Vertical air temperature difference between 1.7 m and 0.1 m above the floor.
b) Mean of three measurements at one position (manikin´s breathing zone) ± 95 % confidence limit.
c) Mean value from one measurement at nine positions (C1, C2 and C3 at 0.6 m, 1.1m and 1.7m) ± standard deviation.

4. DISCUSSION AND CONCLUSION

Referring to Tab. 1, warm air heating and floor heating systems did not show any significant risk of thermal discomfort due to vertical temperature asymmetry or draught in the present study. The effect of increasing nominal air change on the ventilation effectiveness was found to depend on the position of the air terminal devices, and varied between 0.4 and 1.2, where 1 is complete mixing. When a radiant floor heating system was used, the necessarily cooler supply air mixed well and created very uniform conditions with a ventilation effectiveness close to 1.

From Tab. 2 it can be seen that the floor cooling did not show any significant risk of thermal discomfort due to draught, but problems with vertical air temperature differences may occur mainly when floor cooling is combined with warm fresh air supply. This may no longer correspond to the requirements on comfortable indoor environment. The supply air mixed well when the floor cooling system was combined with cold air supply, but the ventilation effectiveness depended on the position of ATDs and on the nominal air change rate when the air supplied in the room was warmer than the room air, and the ventilation effectiveness varied between 0.5 and 1.3, where 1 is complete mixing.

A number of indoor environment indicators have been used in the present study that allow detailed investigation of the indoor environment, mainly when used for experimental measurements under well-defined boundary conditions. However, to complete the information, the experimental measurements of objective indicators should be supplemented by other means of investigation, as e.g. field measurements, subjective evaluations or by visualisations by CFD simulations (using the experimental data for validation).

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